1954

The effect of vibration on heat transfer by forced convection to a compressible fluid in a horizontal tube

Holden, John Joseph

Monterey, California: U.S. Naval Postgraduate School

http://hdl.handle.net/10945/14127

THE EFFECT OF VIBRATION ON HEAT TRANSFER BY FORCED CONVECTION TO A COMPRESSIBLE FLUID IN A HORIZONTAL TUBE

JOHN JOSEPH HOLDEN

Artisan Gold Lettering & Smith Bindery

593 - 15th Street Oakland, Calif. GLencourt 1-9827

DIRECTIONS FOR BINDING

BIND IN

(CIRCLE ONE)

BUCKRAM

COLOR NO. 8854

FABRIKOID

COLOR ________

LEATHER

COLOR ________

LETTERING ON shelf BACK TO BE EXACTLY AS PRINTED HERE.

HOLDEN

1954

Thesis
H68

OTHER INSTRUCTIONS

Letter in gold. Letter on the front cover:

THE EFFECT OF VIBRATION ON HEAT TRANSFER
BY FORCED CONVECTION TO A COMPRESSIBLE
FLUID IN A HORIZONTAL TUBE

JOHN JOSEPH HOLDEN

THE EFFECT OF VIBRATION ON HEAT TRANSFER

BY FORCED CONVECTION TO A COMPRESSIBLE

FLUID IN A HORIZONTAL TUBE

* * * * * *

J. J. Holden

THE EFFECT OF VIBRATION ON HEAT TRANSFER BY FORCED CONVECTION TO A COMPRESSIBLE FLUID IN A HORIZONTAL TUBE

by

John Joseph Holden,
Lieutenant, United States Navy

Submitted in partial fulfillment
of the requirements
for the degree of
MASTER OF SCIENCE
IN
MECHANICAL ENGINEERING

United States Naval Postgraduate School
Monterey, California

1 9 5 4

This work is accepted as fulfilling

the thesis requirements for the degree of

MASTER OF SCIENCE

IN

MECHANICAL ENGINEERING

from the

United States Naval Postgraduate School

PREFACE

As often occurs in industry, certain standard heat exchanger equipment is subject of induced vibrations by the proximity of other operating machinery. It has been found that such heat transfer units, designed to be stationary, often time performed beyond expectations when vibrations were thus induced and were considered overdesigned. R. C. Martinelli and L. M. K. Boelter (6) initiated a partial investigation of the phenomenon under conditions of free convection from a horizontal cylinder. They conducted their research with the thought towards the possibility of designing a heat exchanger to be vibrated with a consequent reduction in size for the same heat transfer rate. In order to determine the feasibility of this method of accelerating heat transfer, they suggested that an economic study be made in order to find a balance between the decrease in size of such a unit for the same rate of heat transfer and the resulting decrease of life of the equipment.

W. A. Grosetta and R. M. George (4) continued the investigation by determining the effects of vibration on heat transfer to water by forced convection in a horizontal tube. They hypothesized that the vibrations would increase the amount of fluid turbulence near the surface thus increasing the amount of heat transfer by convection.

The equipment used by Martinelli and Boelter consisted of a horizontal cylinder, 0.75 inches in diameter and 12-5/8 inches long. The pipe, or cylinder, was heated internally by electricity and vibrated in a large cylindrical tank of water. The vibrations were varied from

zero to 40 cycles per second and the amplitude from zero to 0.1 inches. Three four-junction thermocouples measured the temperature of the water at distant points in the bath. The procedure consisted of first varying the frequency and amplitude of vibration for constant power inputs and second of varying the power input at constant frequency and amplitude.

Martinelli and Boelter found in their study of heat transfer by free convection that below a critical value of Reynolds and Modulus (about 1,000) the velocities of vibration have no effect on the rate of heat transfer. As the Reynolds Modulus is increased beyond the critical value the rate of heat transfer increases with increasing velocity of vibrations up to a value of Reynolds Modulus of about 7,000, beyond which point the effect of vibrations become negligible.

Grosetta and George found that for low flow rates of water (below about 2,340 lbs./hr.ft^2) in the non-boiling range the effects of vibration had the unexpected effect of decreasing the surface heat transfer coefficient. This decrease apparently was dependent upon frequency and to a lesser degree upon amplitude. As the Reynolds number was increased the effect of vibrations decreased and above a Reynolds number of about 4,500 the effect became small and finally disappeared. Three runs were made near the boiling range of the water and a trend towards an increase of heat transfer was noticed. At this higher heat rate it is probable that vapors were being formed or dissolved gases in the water were being released and existed as compressible bubbles in the stream. It was surmised that the bubbles thus formed brought about an increase of turbulence in the flow of water and thereby caused

an increase of heat transfer. This last trend seems to indicate that it is necessary to have a compressible fluid in order to increase heat transfer with vibrations by forced convection.

It is to be noted that the investigation of Grosetta and George was conducted almost wholly in the laminar and transitional flow regions of water. In this region of flow it is difficult to compare experimental results with existing analytical equations. As pointed out by E. R. G. Eckert (3) this is due principally to three effects, the variation of viscosity with temperature for liquids, the existence of eddy currents at low flow rates, and the long tube length required to fully develop the flow both hydrodynamically and thermally. Referring to figure 6 there seems to be a definite trend of the curves for a Reynolds number below 4,000, however, with a different experimental set-up these same results would probably not be the same. They offered the explanation that a swirl about the tube axis was probably set up by the vibrations causing a flow disturbance which interfered with the natural convection processes. This interference was interrupted as the Reynolds number was increased and the turbulent flow region approached. Such an explanation was very likely the case.

In view of the fact that Grosetta and George found that compressibility seems to be a governing factor it was decided to make a further study of the phenomenon by conducting a comparable investigation of the effects of vibration on heat transfer using a compressible fluid. Air seemed to be the most logical choice of fluid. As most standard air heat exchanger equipment operates in the turbulent flow region, it was further decided that for the study to be of a practical nature

this region should be investigated. Both previous investigations had found that in this region for a non-compressible fluid the effects of vibration were small or indiscernible. Although the investigation could quite possibly produce negative results it was felt that the turbulent flow region should be examined in order to determine if such effects did exist there for a compressible fluid.

The investigation was conducted during the first part of 1954 at the United States Naval Postgraduate School, Monterey, California.

The author is greatly indebted to Professor C. D. G. King for his helpful suggestions and constant guidance during the progress of the investigation.

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

Table I

INDEX OF SYMBOLS

A	Area of heat transfer surface, (ft^2)
a	Amplitude of vibration, (in)
c, c_p	Specific heat of fluid at bulk temperature, (BTU/lb ^oF)
D	Outside diameter (ft)
d	Inside diameter (ft)
Δt	Temperature difference, (^oF)
E	Voltage, (volts)
f	Frequency of vibration, (cps)
G	Mass velocity of fluid flow, (lb/hr ft^2)
h	Surface coefficient of heat transfer, (BTU/hr ft^2 ^oF)
I	Electrical current (amperes)
k	Thermal conductivity of fluid at bulk temperature (BTU/hr ft ^oF)
L	Length of heated test section, (ft)
m	Mass rate of fluid flow, (lb/hr for water, cu ft/min for air)
Nu	Nusselt number, (hd/k)
Pr	Prandtl number, (cu/k)
q	Rate of heat flow, (BTU/hr)
r_i	Inside radius (ft)
r_o	Outside radius (ft)
Re	Reynolds number (dG/u)
t	Temperature (^oF)
t_1	Temperature in tube wall, section 1
t_2	Temperature in tube wall, section 2
t_3	Temperature in tube wall, section 3

t_4	Temperature in tube wall, section 4
t_5	Entrance temperature of the fluid
t_6	Exit temperature of the fluid
t_7	Temperature outside lagging, section 1
t_8	Temperature outside lagging, section2
t_9	Temperature outside lagging, section 3
t_{10}	Temperature outside lagging, section 4
u	Absolute viscosity of fluid at bulk temperature (lb/hr ft)
W	Power, (watts)
$\dot{v}$	Volume rate of flow (cu ft/min)

CHAPTER I

SUMMARY

In determining the effects of vibration on heat transfer to a compressible fluid by forced convection, a thick-walled copper tube, approximately 1/2 inch in diameter and one foot long was employed. The test section was heated externally by wrapping it with an electrical heating coil. Temperatures were determined at four positions along the tube by appropriately located thermocouples. The test section was thermally insulated and mounted on a fatigue vibrator with associated controls for frequency and amplitude adjustment.

Air was chosen as the compressible fluid and the tests were conducted in the turbulent region of flow. A series of runs were made, first without vibrations, and then by vibrating the test section transversely over a range of frequencies and amplitudes.

It was found for the particular experimental arrangement used that vibrations have a negligible effect on heat transfer to air in the turbulent flow region. These results agree with the findings obtained in previous investigations for a non-compressible fluid in the same flow region.

CHAPTER II

DESIGN CONSIDERATIONS

It was determined that the same test section that was utilized by W. A. Grosetta and R. M. George would be satisfactory for further investigation in the region of turbulent flow for air. Consequently the design considerations are the same as those presented in their work. These considerations are discussed briefly with a few modifications it was found desirable to make.

In designing the test section a thick walled copper tube was decided upon. The thick walls of the tube permitted accuracy in locating the thermocouples, permitted them to be installed easily, reduced the distortion of the heat flow pattern caused by the thermocouple wells to a minimum, and added regidity to the test section. The choice of copper for the material of the test section was made because of the metal's high thermal conductivity. This decision proved sound, as it was found that at the most there was only a 1°F. temperature drop across the cross-section of the tube. Because of this small temperature drop and the accuracy of location the thermocouples in deep wells near the inner surface of the tube, it was possible to extrapolate and obtain the inner surface temperature of the tube with only a negligible error.

Near each end of the test section a groove, 1/16 inches wide and 1/8 inches deep was cut around the circumference of the tube to reduce the amount of heat loss out of the end of the tube to a negligible value. The ends of the test section were mounted in bakelite end flanges to further cut down the loss of heat in this direction.

Number 17 Nichrome V electrical resistance wire was chosen as the heating coil with which to wrap the test section. The use of this wire allowed 17 turns of the coil to be wrapped for each linear inch of the test section, which considering the 220 volt power supply gave the maximum amount of heat input without exceeding the temperature limits of the coil, and gave the most even distribution of heat along the test section.

Ordinary steam pipe insulation consisting of 85% magnesia was decided upon as the insulation covering for the outside of the test section, because of its acceptability and availability.

It was decided that the circulation of hot air through the outer jacket did not accomplish the purpose for which it was designed, namely, that of insulating the test section from any radial heat loss. Consequently this system was omitted, and the radial heat loss was to be calculated by location thermocouples radially outward from the test section outside the pipe insulation.

In previous work it was considered that the thermocouple located on the outlet side of the test section did not give reliable indications of the exit temperature of the fluid due to insufficient mixing obtain the bulk water temperature at this location. For this reason a mixing chamber was designed and located immediately behind the test section as it is important to determine this temperature as accurately as possible.

The mixing chamber was designed using a 4 inch section of a 2 inch pipe coupling as the main body. At a distance of 1 3/8 inches from each end slots were cut radially from opposite sides of the coupling 1/16 of an inch thick and extending about 1/4 inches beyond the center

of the coupling, see figure 9. A strip of sheet metal was then silver soldered in place in the slots and the excess trimmed and ground flush with the outside of the coupling. A 2 inch to 1/2 inch bell reducer was put on each end and a standard 1/2 inch pipe coupling attached. The exposed ends of these couplings were machined down to adapt them for a rubber hose connection. Two thermocouples were to be placed after the last baffle leading into the mixing chamber from opposite sides of the 2 inch coupling.

It was found from experience that it was necessary to design a means of protecting the thermocouple leads at the point where they lead from the test section through the heating coil. The intense heat given off by the coil was sufficient to burn the thermocouples off at this point. Small ceramic insulating beads were inserted around the thermocouples at this point to insulate them from the heating coil.

CHAPTER III

ASSEMBLY OF THE TEST SECTION

A diagram of the test section is presented in figure 7. The test section is a 3/4 inch double extra heavy cold-drawn seamless copper tube. It is 12 3/8 inches long with an inner diameter of 0.524 inches and an outer diameter of 1.05 inches. The grooves mentioned in the previous chapter were machined 1/8 inches from each end and are 1/16 inches wide and 1/8 inches deep, making the actual test section 12 inches long. The purpose of these grooves is to reduce the longitudinal heat flow out of the ends of the test section to a negligible amount.

Four thermocouple wells were drilled in the test section in such positions that by dividing the test section into four equal sections, each 3 inches long, each thermocouple well was located at the mid point of a section. The thermocouple wells were drilled with a #56 drill to a depth of 0.200 inches and finished with a flat bottom, using a #56 flat-end drill in order that the thermocouple beads could be accurately located 0.063 inches from the inside surface of the test section.

Thermocouple beads were made as outlined by H. Dean Baker (1) first by the butt-weld junction method using the model J-E-S Micro-Weld Butt Welding machine and secondly by the arc method for forming a beaded junction using a mercury pool. The second method proved to be the easier and by far less time consuming. Both the butt-weld and beaded junctions proved to be satisfactory in use. Number 30 B and S gauge iron-constantan thermocouple wires were used.

The test section was then mounted between the two bakelite end flanges and secured in the flanges by four tie rods. The end flanges

were machined from 2 inch thick pieces of bakelite. This material was chosen for the flanges because of its insulating, heat resistance and mechanical properties. Their purpose besides holding the test section was to further cut down any longitudinal heat flow from the ends of the test section. A V-shaped groove was machined in the end of the flanges to provide a seat for a silicone O-ring gasket. Silicone rubber was selected for the gasket material because of its high temperature resistance properties.

After the test section was mounted between the end flanges and secured in place by the four tie rods, the whole assembly was mounted in the vibration jig assembly and secured in place. The four thermocouple leads were then cemented in the wells using technical G type quick drying cement. A ceramic insulating bead was placed around each thermocouple lead after the bead had been machined down to as small a thickness as was practical.

The test section was then wrapped with the number 17 Nichrome V wire. The wire was insulated from the test section by a double thickness of fiber glass insulation. This made the over-all diameter of the heating coil wire 0.06 inches, allowing 17 turns of the coil per linear inch of the test section, as designed. The fiber glass insulation had been tested in a furnace to a temperature of 1200° F and proved to be very satisfactory. It was very difficult to handle, though, because it frayed easily exposing the wire. This difficulty was eliminated by coating the insulated wire lightly with a thin coat of glyptal insulating varnish. It was then possible to wrap the wire around the test section without any fraying of the insulation. The

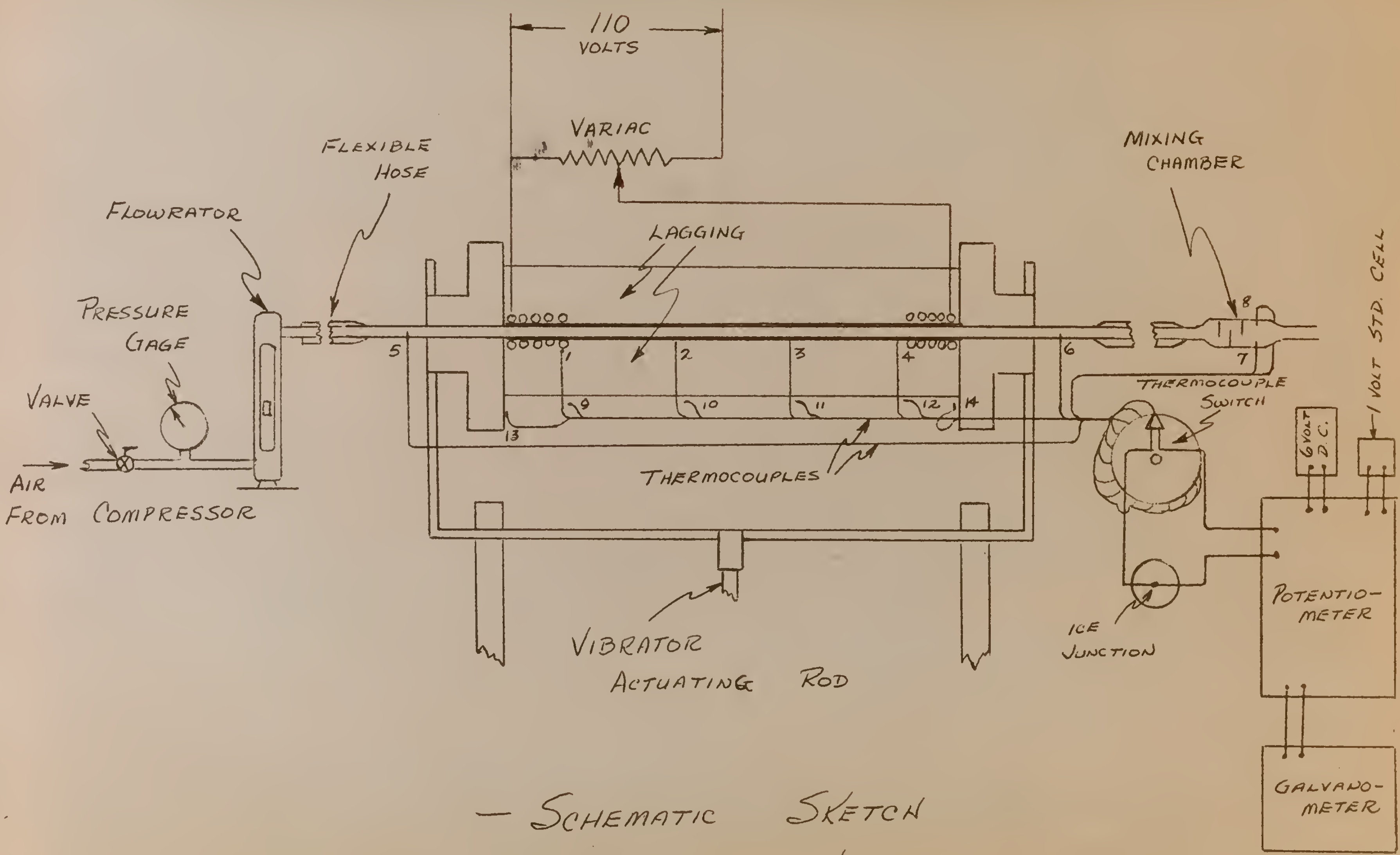
wire was then wrapped in as tight a coil as possible around the test section, thus giving an even distribution of heat. Two of the tie rods were made of brass and served as electrical busses. A short section of #14 copper wire was silver soldered to each end of the heating coil and each was wrapped around a tie rod which served as an electrical bus. Thus little heat was generated outside of the heating coil which was entirely on the 12 inch portion of the test section.

The test section was next carefully wrapped with a single layer of asbestos tape. This helped to cut down the flow of heat radially and to hold the heating coil in place. Standard 85% magnesia pipe insulation was next fitted snugly over the test section in such a manner that the four thermocouple leads from the test section could pass through the pipe insulation where the two halves of the insulation were butted together. The insulation was then secured in place with another wrapping of asbestos tape. Four thermocouple leads were placed outside the pipe insulation just under these wrappings in positions which were on a circumference radially outward from the thermocouples in the test section and displaced slightly from the joint formed by the two halves of pipe insulation. An aluminum sheet cover was then installed around the entire test section assembly.

Copper tube adapters were threaded into the outer annulus of the end flanges. Thermocouples were located in each of the adapters in order to determine the entering and leaving temperature of the fluid. These thermocouples extended into the flow region of the fluid.

The mixing chamber was built as designed. It was then connected to the outlet annulus by a short length of rubber hose. The connection

assembly from the test section to the mixing chamber was then completely wrapped with numerous layers of asbestos tape in order to cut down on any heat loss between the two units. The purpose of the mixing chamber was to obtain a complete mixing of the leaving fluid in order to obtain its average bulk temperature. This mixing chamber proved to be quite successful when used with water and indicated that the bulk water temperatures were anywhere up to 1 1/2 $^{\circ}$F above that temperature measured by the thermocouple in the exit tube adapter. With air, however, it was found that by the time the air had passed through the mixer the heat loss was sufficient to drop its temperature below that indicated by the thermocouple in the exit tube adapter. Thus, while the temperatures measured in the mixing chamber were used in making the calculations for water, they were not used in making those for air and the temperatures recorded by the exit tube adapter thermocouple were used.



— SCHEMATIC SKETCH OF TEST LAYOUT —

FIG. 1

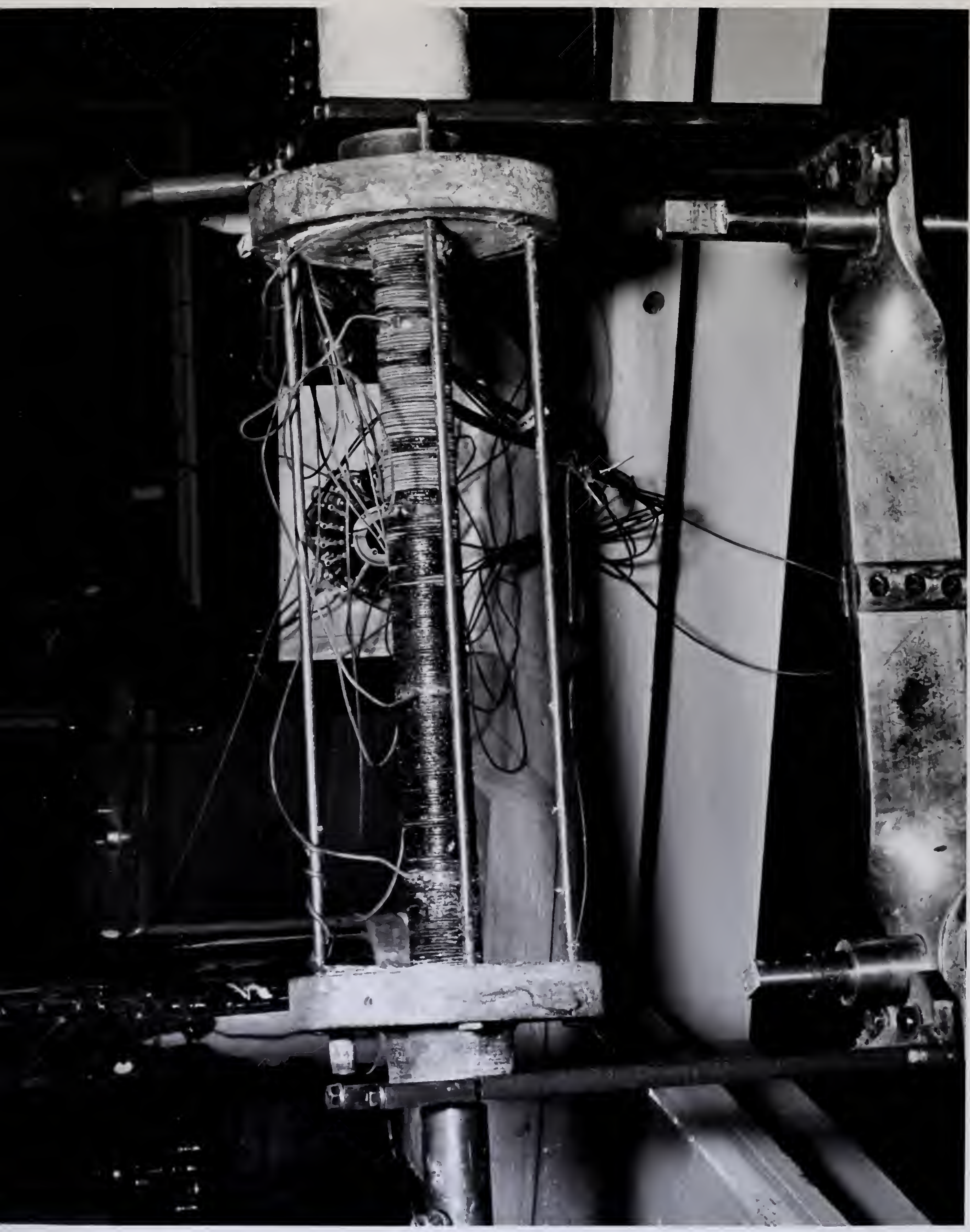


Figure II Photograph of the Test Section

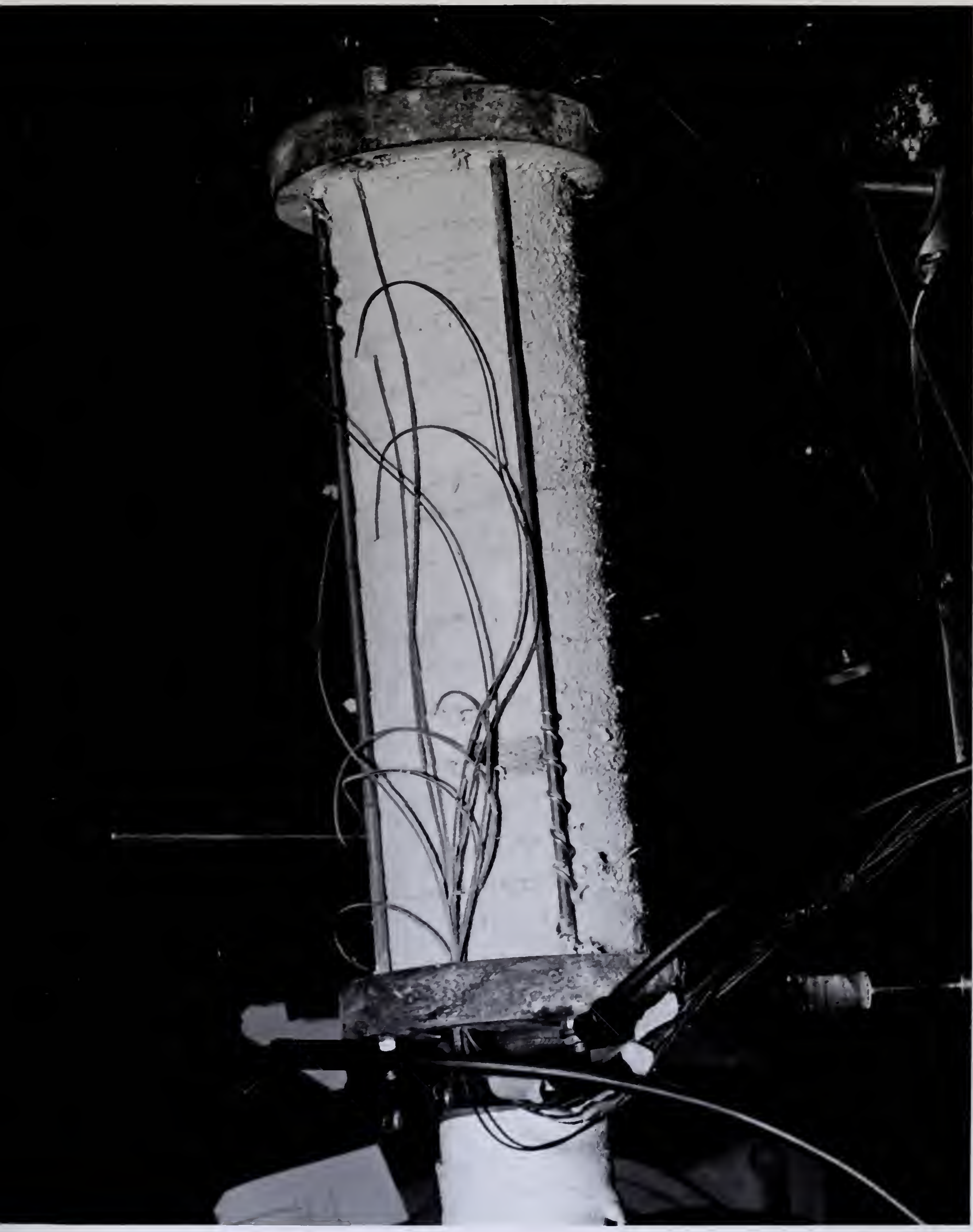


Figure III Photograph of the Test Section with Lagging

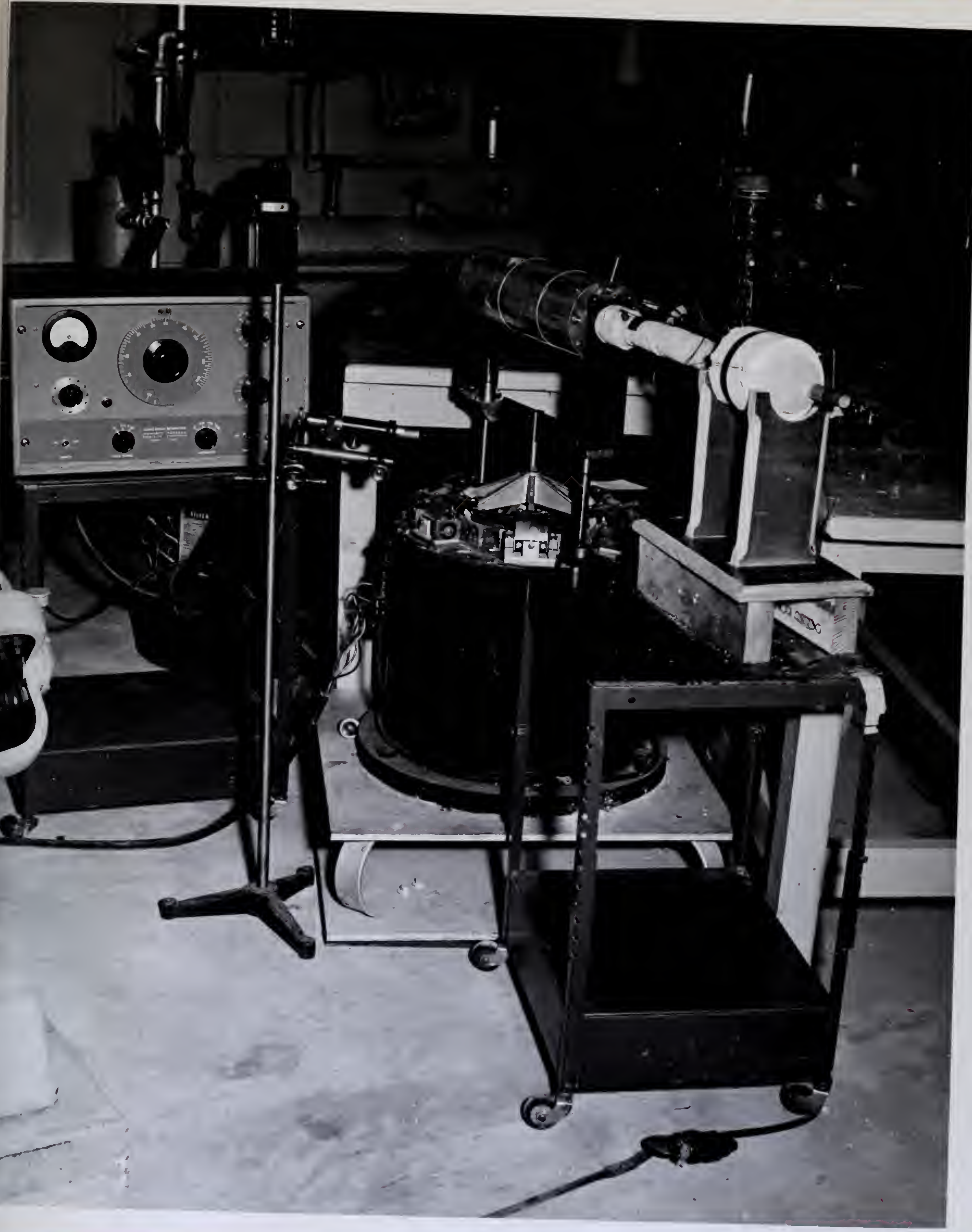


Figure IV Photograph of the Vibration Assembly

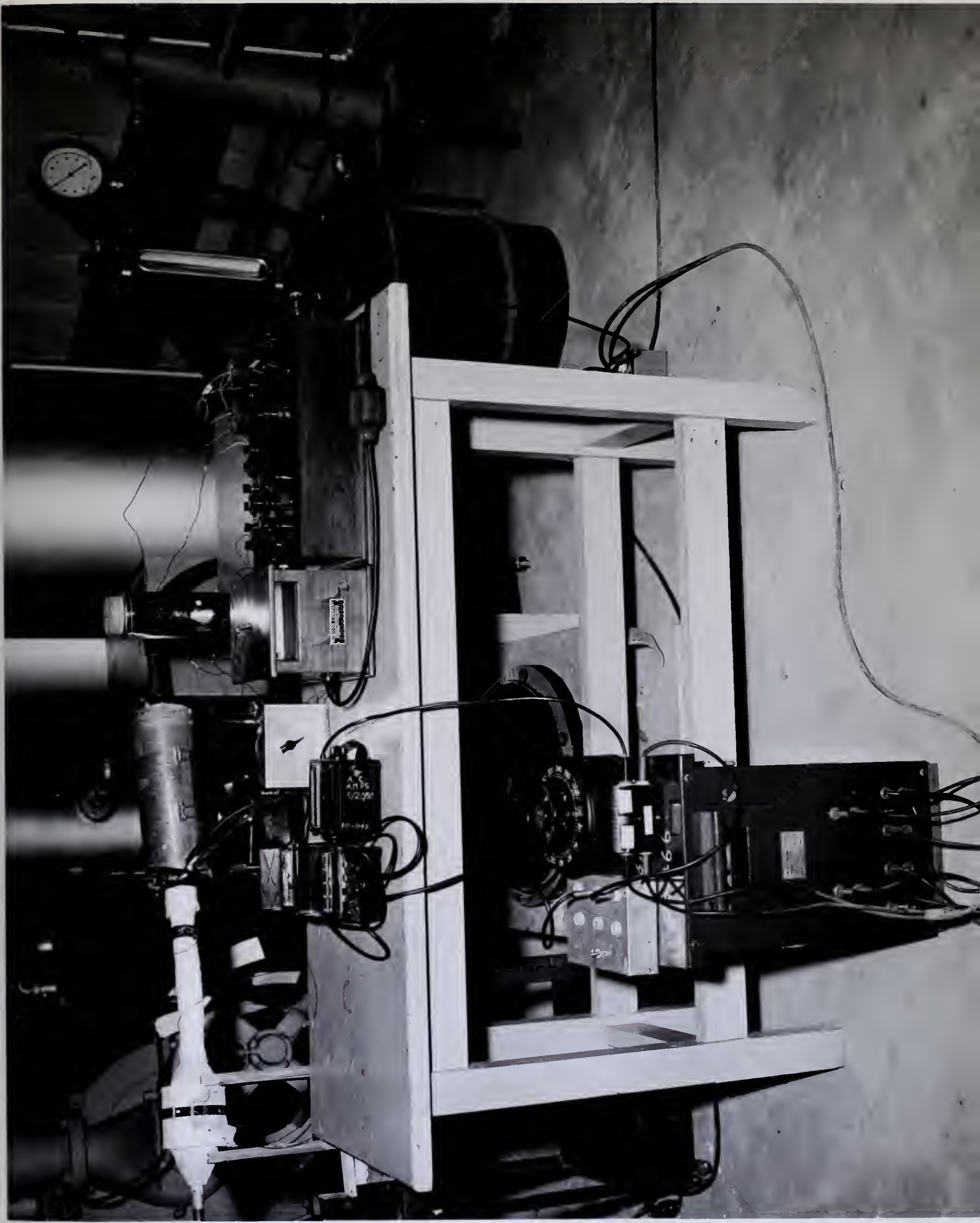


Figure V   Photograph of the Thermometry Apparatus

CHAPTER IV

ASSOCIATED TEST EQUIPMENT

The associated test equipment used in this investigation consisted of the Westinghouse vibration fatigue machine and its associated equipment, the external power circuit, the temperature measurement equipment and the air supply and flow measurement devices.

The Westinghouse vibration fatigue equipment included the Westinghouse type HI 40 amplifier-exciter, the HI 500 power amplifier and voltage regulator and the HI vibration motor. A Hewlett Packard model 205 A audio signal generator was used in conjunction with this equipment.

The test section was mounted in a vibration jig assembly which in turn was mounted on the drive coil and crown assembly of the vibration motor. The vibration jig assembly consisted of two vertical guide rods which were attached to the body of the vibration motor. A cross bar which could move freely up and down was held horizontally and guided by the vertical rods. A driving rod connected the cross bar to the drive coil and crown assembly of the vibration motor and transmitted the vibrations of the motor to the cross rod. The test section was in turn securely connected to the cross rod through two vertical clamps which held the bakelite flanges of the test section. Rather critical adjustment of the two vertical guide posts was required in order that the amplitude of the vibrations would not be damped. It was possible to obtain vibrations of the test section from 20 to 20,000 cycles per second. At low frequencies fairly large amplitudes of vibration could be obtained, but at frequencies of above 60 cycles per second the

amplitude was greatly cut down. It was noted that at frequencies of above 40 cycles per second the vibrations were not altogether vertical but also included a horizontal component. The amplitudes of vibration were measured by a traveling microscope.

The external power circuit supplying the heating coil consisted of a 220 volt alternating current power supply which was controlled by a 0-235 volt variac. Measuring instruments were standard and consisted of an ammeter, 0-5 and 0-20 amperes, a voltmeter, 0-150 and 0-300 volts, and a wattmeter, 0-5 and 0-50 watts.

The temperature measuring equipment consisted of 14 thermocouples, a switching device, a vacuum thermos ice-point, and a standard potentiometer and sensitive galvanometer.

The air supply consisted of a 100 pound per square inch air compressor, four storage tanks, a filter section filled with silica-gel for drying the air. The flow of air was metered by a Porter and Fisher series 700 flowrator meter. A standard 0-30 psig pressure gauge was installed immediately after the flow control valve in order to measure the air pressure. The flowrator meter was checked for a series of runs using water and found to give accurate flow rates. The calibrated orifice method of flow measurement used previously was abandoned due to its relative inaccuracy.

CHAPTER V

PROCEDURE

At the commencement of a run the frequency and amplitude of vibration were first adjusted. The desired rate of flow of air through the test section was obtained and the proper voltage adjustment was then made. After about two hours the thermocouples at the outside of the lagging would indicate constant temperature and that steady state conditions had been reached. The voltage, current, power and flow rate were then recorded. The thermocouple readings were quickly taken followed by a check of the voltage, current, power and flow rate readings to ascertain that they were the same. If so, the frequency and/or amplitude of vibration were adjusted to the next desired value and the run continued in the same manner.

For each run the power input to the heating coil and the rate of flow of the air were maintained constant. The readings were recorded, first without vibrating the test section, and then for the various combinations of frequency and amplitude desired.

The first four runs were made using water as the fluid and without vibrating the test section in order to ascertain agreement with the results obtained by Grosetta and George for the same input conditions and to compare the results with curves presented by W. H. McAdams (7). The remaining runs were made using air as the fluid.

CHAPTER VI

RESULTS

Referring to the results of the calculations (Appendix I) it is seen that the variation of the surface coefficient of heat transfer (i.e., h) is negligible for each run. The slight variation that does exist decreased with increasing frequency of vibration in most cases, however, the total variation of h is about the same as the range of experimental error for the test equipment.

In order to determine the effect on the heat transfer coefficient of a possible error in the temperature readings of the tube wall, an error of $\neq 1$ °F was assumed for run 10-a. The value of h using the recorded wall temperature was 38.0 BTU/hr ft^2 °F. A change of the tube wall temperature of $\neq 1$ °F gave a value of 38.4 BTU/hr ft^2 °F. The overall change of h over the range of frequencies for a run in most cases is only about two or three times as large as this slight variation for as error of 1 °F. The variation in h found when considering an error in temperature measurement was of such a magnitude that it may have masked the slight variation of the actual value of h.

The results obtained for each run in the non-vibrating condition are plotted in figure 6 to coordinates of $Nu/(Pr)^{0.4}$ versus Re. Values obtained from the Colburn equation (7), namely,

$$hd/k = 0.023 \ (dG/u)^{0.8} \ (cu/k)^{0.33}$$

are also plotted. This equation gives good correlation with experimental data for fluids of high viscosity and Reynolds numbers exceeding 10,000. The experimental results obtained from this investigation agree within 20% of the curve obtained from this equation. The results

obtained by Grosetta and George are also plotted in order that the reader may see the effects of vibration on heat transfer for a non-compressible fluid.

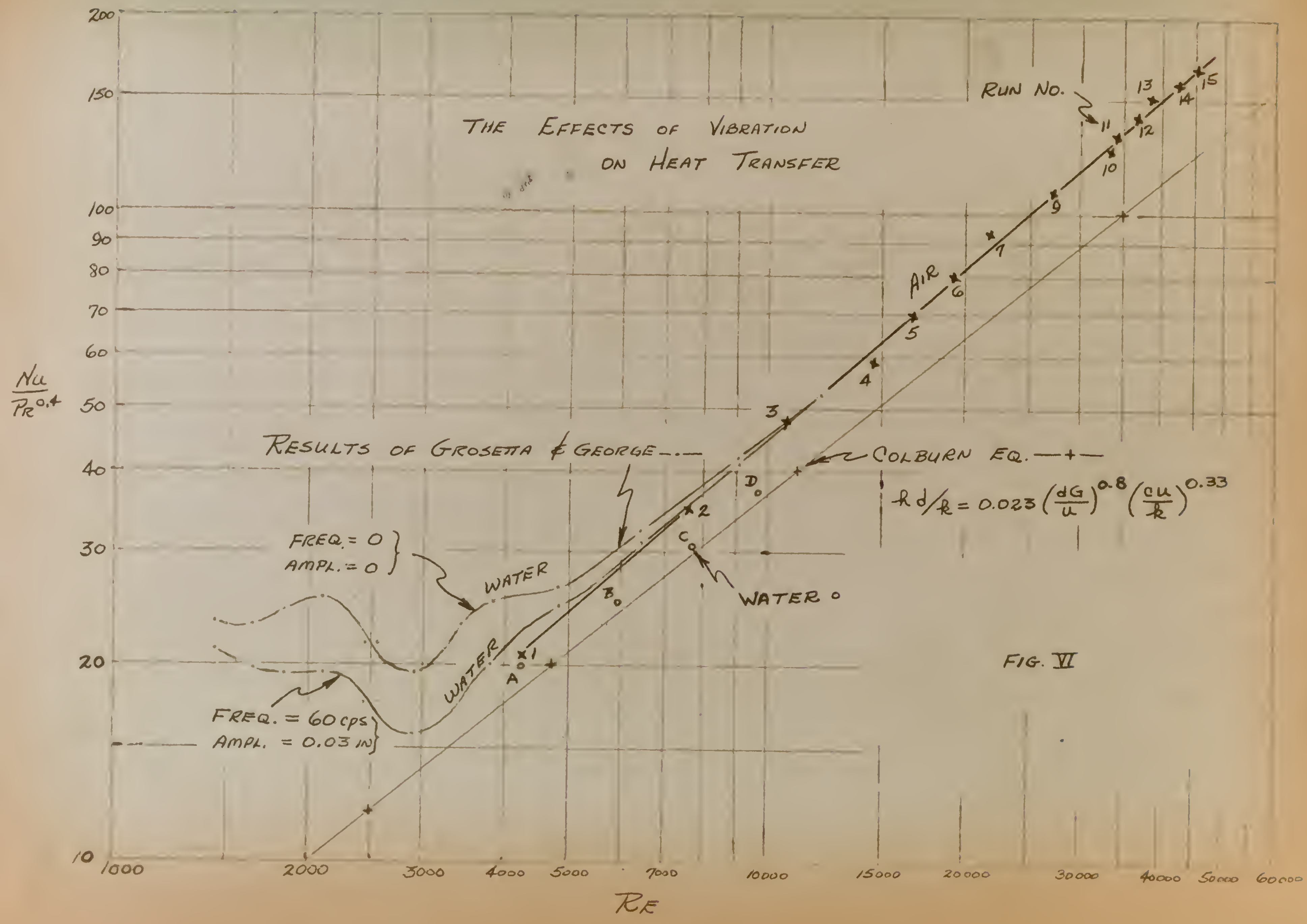

THE EFFECTS OF VIBRATION
ON HEAT TRANSFER
RUN NO.
1
2
3
4
5
6
7
9
10
11
12
13
14
15
AIR
RESULTS OF GROSETTA & GEORGE
COLBURN EQ.
$h d/k = 0.023 \left(\frac{dG}{u}\right)^{0.8} \left(\frac{cu}{k}\right)^{0.33}$
FREQ. = 0
AMPL. = 0
WATER
WATER
WATER o
FREQ. = 60 cps
AMPL. = 0.03 IN
A
B
C
D
$\frac{Nu}{Pr^{0.4}}$
10
20
30
40
50
60
70
80
90
100
150
200
1000
2000
3000
4000
5000
7000
10000
15000
20000
30000
40000
50000
60000
RE

FIG. VI

CHAPTER VII

CONCLUSIONS

It may be concluded from the results obtained from this investigation that for the particular experimental equipment arrangement used vibrations of the heating surface have a negligible effect on the heat transfer coefficient to air flowing turbulently during forced convection. These results are substantially the same as those obtained by Martinelli and Boelter and Grosetta and George for a non-compressible fluid, i.e., that above a Reynolds number of about 4,500 the effects of vibration are negligible.

There are indications from the data obtained that there may be a very slight effect, but more elaborate test equipment and instrumentation would be required to detect this change. As the effects are almost negligible it would be uneconomical to design a heat transfer unit for fluids in the region investigated with the thought towards improving heat transfer by vibrating the unit.

The phenomenon that takes place during turbulent flow through a pipe has been presented by E. R. G. Eckert (3). For this type flow there always exists a laminar sublayer adjacent to the inner surface of the pipe. If the thickness of this sublayer is represented by the amount $d(y)$, then the amount of heat flow through the layer by conduction is, $q = -k\, d(t)/d(y)$. When the turbulent flow region for a fluid is first entered it seems safe to surmise that this sublayer is relatively thick, but that as the mass rate of flow is increased the thickness decreases rapidly, probably asymptotically, to a small thickness, but never quite disappears. Assuming that this is the case,

it seems plausible that transverse vibrations would increase the volume of the turbulent region, thereby decreasing the thickness of the laminar sublayer. This decrease of thickness would necessarily be small, particularly at higher mass rates of flow, but any decrease of d(y), however slight, should increase proportionally the amount of heat passing through the sublayer.

For laminar flow it seems logical that induced vibrations would introduce turbulance along the axis of flow. In this case, the laminar sublayer, which literally extended across the cross sectional area of the tube before vibrations, was reduced considerably in thickness, especially when compared with the reduction in thickness that occurred when vibrating the turbulent flow case. It would be expected, therefore, that vibrations would have a much more pronounced effect on heat transfer for the laminar flow condition than for the turbulent flow, which is the case.

It should be noted at this point that the tube length plays an important part in considering the effects that take place. The length of tube required for the temperature and velocity profiles to be fully developed in a uniformly heated tube may be calculated from the equation,

$L/d \doteq 0.05(Re)(Pr)$, see reference (3) page 103. From this equation, the tube length required for the temperature profile to be developed for air flowing in a 1/2 inch diameter tube and having a Reynolds number (Re) of 4280 and a Prandtl number (Pr) of 0.873, (i.e., run 1) would necessarily be over 8 feet. For water for a corresponding Reynolds number the tube would have to be over 70 feet long. It was shown by Nusselt (3) that the Nusselt number, Nu = hd/k, starts

with an infinite value at the beginning of a heated length of tube and decreases asymptotically to a constant value depending upon the fluid.' For air with a Reynolds number of 4,280 this requires a length of tube of over 2 feet, for the Nusselt number to reach a constant value.

For practical purposes the investigation was necessarily conducted using a much shorter test section than that required. As the Nusselt number, and thus h, and the temperature profile were not fully developed for the length of tube used, but were in a state of change, it is quite possible that h could decrease with vibrations in both the laminar and turbulent flow regions for either a compressible or noncompressible fluid.

CHAPTER VIII

RECOMMENDATIONS

It is recommended that any further investigations of heat transfer by forced convection under conditions of vibration be conducted with certain modifications, namely,

1. An investigation using a test section with a smaller inner diameter in order to have a better L/d ratio.
2. An investigation using a fluid as wet steam or a mixture of air and water in the laminar and turbulent flow regions.
3. An investigation in the laminar flow regions using a compressible fluid as air.
4. An investigation correlating the effects of vibrating the test section in a direction parallel to rather than transversally to the direction of flow of the fluid.

BIBLIOGRAPHY

1. Baker, H. Dean. Manual on Thermometry. East Hartford, Conn.: United Aircraft Corporation, 1950.

2. Baker H. Dean, E. A. Ryder, and N. H. Baker. Temperature Measurement in Engineering. Vol. I. New York: John Wiley & Sons, 1953.

3. Eckert, E. R. G. Introduction to the Transfer of Heat and Mass. New York: McGraw-Hill, 1950.

4. Grosetta, W. A., and R. M. George. The Effect of Vibration on Heat Transfer by Forced Convection in a Horizontal Tube. Monterey, California: United States Naval Postgraduate School, 1953.

5. Jakob, M., and G. A. Hawkins. Elements of Heat Transfer and Insulation. New York: John Wiley & Sons, 1952.

6. Martinelli, R. C., and L. M. K. Boelter. "The Effect of Vibration on Heat Transfer by Free Convection from a Horizontal Cylinder." Proceedings of the Fifth International Congress for Applied Mechanics. Cambridge, Mass.: John Wiley & Sons, New York, 1939, pp. 578 - 584.

7. McAdams, W. H. Heat Transmission. New York: McGraw-Hill, 1942.

APPENDIX I

TABLE II
RECORDED DATA

Run	f	a	m	E	I	t_1	t_2	t_3	t_4	t_5	t_6	t_7	t_8	t_9	t_{10}
WATER AS FLUID			Lb/Hr												
A	0	0	340	135	7.4	122.3	141.0	143.5	146.6	66.5	75.8	135.7	140.8	156.1	159.1
B	0	0	473	135	7.4	106.6	122.1	121.3	121.7	67.4	74.3	131.7	133.2	151.7	157.8
C	0	0	619	135	7.4	102.8	117.0	115.1	114.2	69.0	74.5	133.5	142.1	162.1	178.8
D	0	0	779	135	7.4	97.3	109.5	108.0	106.5	68.5	73.0	133.7	139.1	160.2	178.0
AIR AS FLUID			C.F.M.												
1a	0	0	1.60	52.0	2.80	462.3	490.9	513.0	521.0	78.0	276.4	206.5	207.5	211.5	207.5
b	20	.0252				460.0	489.0	511.7	519.7	75.2	273.6	204.5	205.1	209.9	205.9
c	20	.0441				460.6	489.9	512.4	520.6	71.9	272.0	200.4	201.5	208.4	202.8
d	30	.0252				459.9	489.1	511.5	519.7	73.8	272.8	200.1	201.3	207.6	203.6
e	40	.0220				460.0	490.8	513.3	521.5	76.3	274.8	195.9	204.3	208.5	204.2
f	60	.0126				459.4	490.4	513.2	521.4	74.9	274.1	203.3	204.4	208.7	204.5
g	100	.0047				459.2	490.4	513.2	521.5	74.2	273.4	198.3	202.4	206.5	202.7
h	200	.0047				458.3	489.5	512.2	520.4	73.8	273.0	197.9	200.5	205.9	202.3
2a	0	0	2.65	60.0	3.275	440.7	476.8	503.8	513.7	78.1	259.1	203.7	207.0	215.2	211.3
b	20	.0252				436.6	476.5	503.8	513.7	74.5	256.9	197.1	200.0	207.8	204.5
c	20	.0473				437.6	477.1	504.4	514.5	75.3	257.7	199.0	201.7	211.0	206.6
d	30	.0315				442.8	482.2	510.0	520.4	75.5	260.9	199.7	202.6	212.9	207.6
e	60	.0110				443.6	483.3	511.3	521.3	74.0	258.9	198.6	201.8	212.3	206.4
3a	0	0	3.60	68.0	3.65	432.0	474.7	505.9	517.6	74.5	249.7	201.3	203.4	211.0	206.3
b	20	.0252				430.9	474.0	505.6	517.8	70.5	247.3	201.0	202.8	213.3	206.6
c	20	.0473				429.4	472.7	504.2	516.2	70.8	246.4	202.0	203.4	214.0	207.8
d	40	.0189				431.5	475.0	506.5	518.5	70.9	247.3	200.1	202.5	214.1	207.6
e	60	.0125				430.5	473.8	505.2	517.2	71.3	247.2	199.1	201.7	214.1	207.7

TABLE II

RECORDED DATA

Run	f	a	m	E	I	t_1	t_2	t_3	t_4	t_5	t_6	t_7	t_8	t_9	t_{10}
4a	0	0	5.05	60.0	3.30	298.3	329.3	352.0	360.0	73.6	180.2	148.1	152.8	167.5	167.6
b	20	.0252				295.8	327.4	350.3	359.2	71.2	180.1	144.8	149.2	163.0	160.6
c	20	.0536				295.6	326.7	349.3	357.7	72.0	178.3	144.5	150.9	164.1	161.5
d	60	.0173				304.0	333.9	356.0	364.3	72.7	181.6	145.4	153.1	166.6	167.3
5a	0	0	5.55	74.0	4.02	384.7	427.5	459.6	472.2	68.9	212.5	188.0	190.7	201.5	196.9
b	20	.0252				384.4	427.2	459.5	472.1	65.3	209.3	184.5	188.9	200.6	196.1
c	20	.0441				379.1	421.7	453.9	466.6	63.3	206.2	179.2	181.6	194.6	191.2
d	25	.0400				384.4	419.1	451.4	462.5	62.4	204.4	176.7	179.5	191.1	188.3
e	30	.0299				379.8	417.5	449.7	462.8	61.7	203.1	174.8	177.0	182.8	179.2
f	40	.0299				382.3	420.1	452.3	465.4	61.4	204.2	174.1	177.6	186.1	180.5
g	60	.0112				381.3	437.4	468.9	481.5	79.1	224.3	193.9	199.9	210.4	208.0
h	100	.0047				-	438.4	470.6	483.7	77.3	222.6	192.3	196.9	207.6	204.6
i	200	.0047				-	436.0	477.1	477.1	78.0	222.1	194.2	198.0	209.1	202.6
6a	0	0	6.50	81.0	4.38	390.3	437.7	472.6	488.5	75.5	222.0	199.2	202.6	217.4	218.3
b	20	.0252				390.4	437.7	473.0	488.8	76.1	222.8	198.4	201.7	216.9	217.7
c	20	.0441				389.3	437.2	472.7	488.3	76.7	223.3	197.1	200.4	216.3	217.2
d	40	.0158				390.8	438.2	473.3	488.7	76.4	222.3	194.1	198.4	213.7	215.2
e	60	.0126				387.2	434.8	470.1	485.8	76.5	220.5	195.2	198.9	215.3	215.7
7a	0	0	7.50	81.0	4.38	355.5	401.2	434.1	449.9	76.6	205.1	188.1	193.0	208.8	207.7
b	20	.0252				357.5	403.0	436.1	451.3	78.9	207.0	189.5	194.0	210.0	209.4
c	20	.0472				359.0	404.5	437.8	453.1	79.9	208.5	189.9	194.0	210.4	209.6
d	40	.0189				358.2	403.6	436.8	451.9	80.0	208.2	189.4	195.5	211.1	210.1
e	60	.0126				359.9	405.5	439.0	454.3	80.1	209.0	189.6	195.0	210.8	209.4

TABLE II

RECORDED DATA

Run	f	a	m	E	I	t_1	t_2	t_3	t_4	t_5	t_6	t_7	t_8	t_9	t_{10}
9a	0	0	9.55	86.0	4.70	349.3	398.6	433.3	447.7	82.1	197.9	182.2	186.3	199.4	199.7
b	20	.0629				348.8	397.8	433.1	448.0	83.8	199.0	185.2	188.8	202.1	199.1
c	20	.0315				349.3	396.7	431.4	446.0	83.2	198.3	186.3	190.5	203.7	206.9
10a	0	0	10.60	86.0	4.70	317.5	365.4	399.1	413.2	74.0	179.2	172.6	176.4	190.2	192.4
b	20	.0284				316.8	364.7	398.4	412.6	74.4	179.2	172.6	176.1	188.8	191.2
c	20	.0502				315.4	363.1	396.8	410.9	73.4	177.7	171.1	172.2	186.4	185.1
d	60	.0126				315.4	363.3	397.0	411.4	73.2	177.7	172.1	174.4	188.6	191.3
e	60	.0253				314.9	363.0	396.5	411.0	72.9	177.2	170.3	171.9	186.9	189.8
f	25	.0377				312.9	360.2	394.0	408.3	72.2	176.0	169.7	170.3	187.1	185.5
g	30	.0377				314.0	361.4	395.4	409.5	72.1	176.5	169.2	170.0	186.6	184.7
h	40	.0377				313.3	360.7	394.2	408.5	72.2	176.0	165.4	166.7	181.7	184.3
i	100	.0110				314.2	362.2	395.1	409.4	72.2	176.5	170.2	175.2	190.3	195.0
j	200	.0110				313.1	360.8	394.1	408.5	72.2	176.4	172.7	175.7	190.6	194.6
k	300	.0032				314.2	362.0	395.4	409.6	71.7	176.4	173.4	176.4	191.0	194.8
l	500	.0032				310.0	359.4	392.8	406.9	72.3	172.8	173.2	176.0	190.8	194.6
11a	0	0	11.60	86.0	4.70	299.0	345.4	377.3	391.0	75.7	170.3	170.4	174.3	187.1	188.7
b	20	.0283				296.5	342.7	374.5	388.3	73.7	168.5	167.9	171.3	183.5	186.8
c	20	.0546				296.1	342.2	373.8	387.6	73.3	168.1	166.2	168.8	181.3	180.9
d	60	.0189				293.4	339.6	371.0	385.0	73.4	167.3	165.1	166.2	179.7	183.1
12a	0	0	12.63	96.0	5.10	334.9	390.6	427.9	444.0	78.9	187.0	192.4	196.6	212.5	217.4
b	20	.0283				334.5	390.3	427.7	444.4	79.2	187.7	188.1	191.3	206.7	208.8
c	20	.0536				333.5	389.0	427.0	443.8	77.4	185.5	184.1	186.9	202.4	205.0
d	60	.0188				331.1	387.0	424.7	441.3	76.3	183.2	183.7	185.8	200.3	202.6
e	60	.0310				331.6	387.7	425.2	442.2	77.3	184.3	181.9	182.6	198.2	202.7

TABLE II

RECORDED DATA

Run	f	a	m	E	I	t_1	t_2	t_3	t_4	t_5	t_6	t_7	t_8	t_9	t_{10}
13a	0	0	13.65	96	5.20	315.4	368.6	405.3	420.8	78.3	176.8	179.6	183.3	199.9	196.7
b	20	.0283				315.9	369.8	406.3	422.0	78.6	177.5	179.2	183.1	197.1	197.8
c	20	.0440				315.6	369.5	405.5	421.3	78.7	177.2	179.1	183.1	196.8	197.4
d	60	.0190				314.5	368.3	404.2	420.1	78.7	176.8	179.5	181.5	199.1	204.6
e	60	.0300				314.3	368.2	404.3	420.1	78.8	176.7	178.0	179.5	198.3	204.6
14a	0	0	14.70	96	5.20	297.5	349.0	383.1	397.8	79.7	169.1	176.5	178.8	195.7	201.3
b	20	.0268				296.8	348.6	382.6	397.6	78.7	168.4	173.2	178.4	194.4	200.6
c	20	.0488				294.0	345.7	380.2	395.2	73.1	163.7	162.3	178.0	194.9	200.1
d	25	.0204				294.8	346.4	380.3	395.1	77.2	166.5	171.4	177.3	195.2	201.0
e	60	.0422				301.1	353.3	387.6	402.6	79.3	170.1	173.1	174.1	187.7	189.6
f	60	.0157				300.6	352.3	386.1	400.6	79.0	168.8	178.0	176.6	189.0	187.6
15a	0	0	15.70	96	5.20	284.4	334.4	366.5	381.2	80.6	163.6	175.6	180.1	194.2	200.0
b	20	.0284				286.5	336.2	368.4	382.7	80.5	164.5	174.9	179.2	195.4	201.3
c	20	.0488				281.8	330.2	362.6	376.6	77.6	160.7	169.7	177.3	193.0	199.1
d	60	.0160				287.1	336.7	368.7	383.3	82.6	166.4	177.6	182.0	197.3	203.2
e	60	.0255				286.6	335.8	367.9	382.2	82.7	165.9	176.8	180.9	195.8	201.0
f	25	.0378				279.3	329.1	361.2	375.5	75.5	157.9	170.4	174.9	194.0	199.7
g	30	.0300				281.6	331.9	364.0	378.5	75.7	159.3	173.6	177.9	197.3	203.2
h	40	.0394				284.1	334.1	366.1	380.6	80.1	162.1	175.3	177.5	197.7	204.6
i	100	.0063				285.2	335.5	367.3	382.0	80.4	163.3	175.8	179.6	197.8	204.8
j	200	.0039				285.9	336.3	368.3	382.9	80.5	163.5	173.3	178.0	192.5	200.4
k	300	.0126				286.2	339.0	369.0	383.3	80.3	163.4	174.0	177.3	189.0	189.0
l	500	.0150				285.9	336.5	368.7	383.2	79.8	163.3	170.5	174.6	188.4	189.1

APPENDIX I

TABLE III

REDUCED DATA

Run	f	a	k		u	q	h_1	h_2	h_3	h_4	Nu	Pr	$Nu/Pr^{.04}$	Re
WATER AS FLUID														
A	0	0	.344	1.0	2.33	818	433	351	333	325	42.8	6.8	20.0	4260
B	0	0	.344	1.0	2.33	667	539	366	391	405	54.0	6.8	25.1	5930
C	0	0	.344	1.0	2.33	693	645	432	474	508	65.3	6.8	30.6	7750
D	0	0	.344	1.0	2.33	716	780	525	570	621	79.1	6.8	35.6	9750
AIR AS FLUID														
1a	0	0	.0157	.0707	.0462	338	6.85	6.69	6.86	7.59	18.2	.712	20.85	4280
b	20	.0252					6.85	6.65	6.83	7.54				
c	20	.0441					6.68	6.50	6.71	7.29				
d	30	.0252					6.74	6.55	6.73	7.46				
e	40	.0220					6.80	6.57	6.74	7.42				
f	60	.0126					6.85	6.59	6.71	7.42				
g	100	.0047					6.73	6.53	6.67	7.34				
h	200	.0047					6.74	6.52	6.53	7.39				
2a	0	0	.0156	.0709	.0460	510	10.92	10.59	10.80	11.84	30.6	.712	35.0	7600
b	20	.0252					10.90	10.33	10.79	11.57				
c	20	.0473					10.91	10.40	10.62	11.61				
d	30	.0315					10.68	10.20	10.43	11.38				
e	60	.0110					10.57	10.07	10.40	11.20				
3 a	0	0	.0150	.0715	.0458	674	14.65	13.89	14.00	15.16	42.6	.712	48.1	10570
b	20	.0252					14.50	13.79	13.92	15.05				
c	20	.0473					14.65	13.86	14.02	15.14				
d	40	.0189					14.58	13.73	13.90	15.02				
e	60	.0125					14.52	13.80	13.98	15.11				

TABLE III

REDUCED DATA

Run	f	a	k		u	q	h_1	h_2	h_3	h_4	Nu	Pr	$Nu/Pr^{.4}$	Re
4a	0	0	.0153	.0727	.0452	508	17.52	16.48	16.57	17.94	50.0	.712	57.2	14850
b	20	.0536					17.55	16.51	16.61	17.89				
c	20	.0252					17.51	16.47	16.56	17.80				
d	60	.0173					16.78	16.75	16.17	17.20				
5a	0	0	.0153	.0727	.0452	862	21.1	19.65	19.36	20.6	60.1	.712	69.0	16620
b	20	.0252					20.8	19.42	19.11	20.2				
c	20	.0441					21.0	19.48	19.28	20.4				
d	25	.0400					20.4	19.60	19.30	20.7				
e	30	.0299					20.8	19.65	19.20	20.3				
f	40	.0299					20.4	19.49	19.09	20.1				
g	60	.0112					20.5	19.68	19.45	20.7				
h	100	.0047					20.4	19.40	19.10	20.5				
i	200	.0047					20.3	19.66	19.40	20.9				
6a	0	0	.0155	.0718	.0456	1020	25.1	23.4	23.2	24.5	70.6	.712	80.8	19050
b	20	.0252					25.1	23.5	23.2	24.5				
c	20	.0441					25.2	23.5	23.2	24.7				
d	40	.0158					25.1	23.4	23.1	24.5				
e	60	.0126					25.4	23.4	23.3	24.7				
7a	0	0	.0155	.0720	.0455	1029	28.5	26.2	25.8	27.3	80.2	.712	91.9	21850
b	20	.0252					28.6	26.3	25.8	27.3				
c	20	.0473					28.5	26.2	25.7	27.3				
d	40	.0189					28.6	26.3	25.8	27.4				
e	60	.0126					28.4	26.2	25.6	27.1				

TABLE III

REDUCED DATA

Run	f	a	k		u	q	h_1	h_2	h_3	h_4	Nu	Pr	$Nu/Pr^{.4}$	Re
9a	0	0	.0156	.0715	.0458	1164	33.6	30.0	29.0	30.3	93.6	.712	107.5	27300
b	20	.0629					33.9	30.4	29.2	30.4				
c	20	.0315					33.7	30.4	30.4	30.7				
10a	0	0	.0153	.0727	.0452	1198	38.0	33.8	32.3	33.6	108.3	.712	124.2	33400
b	20	.0284					38.2	33.9	32.4	33.7				
c	20	.0502					38.2	33.9	32.4	33.6				
d	60	.0126					38.2	33.9	32.4	33.6				
e	60	.0253					38.2	33.9	32.4	33.6				
f	25	.0377					38.4	34.1	32.6	33.9				
g	30	.0377					38.2	33.9	32.6	33.7				
h	40	.0377					38.3	34.0	32.5	33.8				
i	100	.0110					38.2	33.9	32.6	34.0				
j	200	.0110					38.5	34.1	32.7	34.1				
k	300	.0032					38.2	33.8	32.6	33.8				
l	500	.0032					39.0	34.1	33.1	33.8				
11a	0	0	.0153	.0727	.0452	1172	40.4	35.4	33.8	35.0	115.2	.712	132.0	34000
b	20	.0283					40.5	35.6	33.9	35.1				
c	20	.0546					40.4	35.6	33.8	35.0				
d	60	.0189					40.9	35.8	34.2	35.4				
12a	0	0	.0155	.0720	.0455	1452	43.6	37.8	36.3	37.4	123.0	.712	140.8	36600
b	20	.0283					43.7	38.4	37.0	38.5				
c	20	.0536					43.3	37.8	36.0	37.1				
d	60	.0188					43.6	37.7	36.1	37.1				
e	60	.0310					43.6	37.7	36.0	37.1				

TABLE III

REDUCED DATA

Run	f	a	k		u	q	h_1	h_2	h_3	h_4	Nu	Pr	$Nu/Pr^{.4}$	Re
13a	0	0	.0154	.0722	.0453	1430	46.3	41.1	37.8	38.7	131.2	.712	153.3	38100
b	20	.0283					46.1	41.1	37.6	38.6				
c	20	.0440					46.3	41.1	37.7	38.7				
d	60	.0190					46.6	41.2	37.9	39.0				
e	60	.0300					46.6	41.2	37.9	39.0				
14a	0	0	.0154	.0722	.0454	1392	49.1	42.0	39.5	40.7	139.0	.712	159.0	42400
b	20	.0268					49.0	41.7	39.5	40.6				
c	20	.0488					48.2	41.2	39.2	40.2				
d	25	.0204					49.0	42.0	39.5	40.7				
e	60	.0422					48.1	41.0	38.7	39.7				
f	60	.0157					48.2	41.0	38.6	39.7				
15a	0	0	.0154	.0722	.0454	1380	52.0	44.4	41.4	42.4	147.1	.712	167.5	45100
b	20	.0284					51.4	44.0	41.2	42.2				
c	20	.0488					51.7	44.3	41.6	42.8				
d	60	.0160					51.6	44.1	41.3	42.5				
e	60	.0255					51.8	44.2	41.5	42.4				
f	25	.0378					51.8	46.0	41.5	42.5				
g	30	.0300					51.6	45.6	41.3	42.1				
h	40	.0394					51.7	43.9	41.4	42.2				
i	100	.0063					51.6	43.7	41.3	42.0				
j	200	.0039					51.3	43.5	40.8	42.0				
k	300	.0126					51.2	42.9	40.8	41.6				
l	500	.0150					51.1	43.1	40.8	41.6				

APPENDIX II

SAMPLE CALCULATIONS

The sample calculations are made for run 10 a, the non-vibrating condition. The test section was divided into four equal lengths, such that, a thermocouple was located at the center of each section. Where indicated the calculations are those for the first section, that is, the first 3 inches of test section length.

1. Calculation of Δt through the tube wall, considering the total length of the test section.

Outer dia., D = 1.050 inches

Inner dia., d = 0.524 inches

Thickness of tube wall = 0.263 inches

Heat released by the coil, q = EI x 3.413 BTU/hr/watt

$$q = \frac{2\pi kL\Delta t}{\ln (r_o/r_i)}$$

k copper = 217 BTU/hr ft ^oF .... (7)

$$q = 86 \times 4.70 \times 3.413 = \frac{2\pi \times 217 \times 1 \times \Delta t}{\ln (0.525/0.262)}$$

$$\Delta t = \frac{86 \times 4.70 \times 3.413 \times \ln 2.002}{2\pi \times 217} = 0.7\ ^oF$$

thermocouple well depth = 0.200 inches

thickness of thermocouple bead = 0.04 inches

Assuming that the temperature measured by the thermocouple is at the center of the bead, the distance from the center of the bead to the inner surface of the tube is, 0.263 - 0.200 + 0.02 = 0.083 inches

Then, the temperature drop from the thermocouple to the inner surface is:

$$\Delta t' = \frac{0.083}{0.263} \times 0.7 = 0.2\ ^oF$$

2. Calculation of the heat loss through the insulation.

$$q = \frac{2\pi kL\Delta t}{\ln (r_o/r_i)}$$

Outer dia. = 3.50 inches

Inner dia. = 1.19 inches

k for 85% magnesia insulation = 0.043 BTU/hr ft $^{\circ}$F ..(5)

$$q = \frac{2\pi(0.043)(3/12)\Delta t}{\ln(3.50/1.19)} = (0.0624)\Delta t$$

Temperature rise from thermocouple bead to the outer surface of the test section or the inner surface of the insulation is:

$$317.5 + (0.7 - 0.2) = 318.0 \ ^{\circ}F$$

$$\Delta t = 318.0 - 172.6 = 145.4 \ ^{\circ}F$$

$$q = 145.4(0.0624) = 9.06 \text{ BTU/hr}$$

3. Calculation of the heat input to the air, considering the total length of the test section.

$$q = mc_p\Delta t$$

ρ (74.0 $^{\circ}$F) = 0.0745 lbm/cu.ft. ...(7)

c_p (74.0 $^{\circ}$F) = 0.2401 BTU/lbm $^{\circ}$F (assumed constant through section)

$$q = (10.6 \text{ cfm})(60 \text{ min/hr})(0.0745)(0.2401)(179.2 \ ^{\circ}F - 74.0 \ ^{\circ}F)$$

$$= 1198 \text{ BTU/hr}$$

Check of calculations:

Input to heater, q = (86)(4.70)(3.413) = 1379 BTU/hr

Conduction along the test section, $q = kA\frac{\Delta t}{\Delta x}$

$$q = (217) \times \frac{\pi}{4} \times \frac{(1.05^2 - 0.524^2)}{144} \times \frac{(413.2 \ ^{\circ}F - 317.5 \ ^{\circ}F)}{9/12}$$

$$= \frac{(217)\pi \ (1.102 - 0.274)(95.7)}{4 \times 144 \times 0.75} = 125.0 \text{ BTU/hr}$$

Then, heat input to the air equals the heat input minus the heat loss through the insulation minus the heat of conduction, or:

$q = 1379 - 9.06 \times 4 - 125 = 1218$ BTU/hr

Thus, there is an apparent additional heat loss of about 20 BTU/hr, which may be leakage through the end of the test section, accumulation of errors, etc.

4. Calculation of the surface coefficient of heat transfer in the first section of the test pipe. (Based on heat guided by air)

$h = q/A\Delta t$

$$= \frac{1198/4}{\pi(0.524/12)(3/12)\quad t} = \frac{(1198)(12)}{(\pi)(0.524)\Delta t}$$

Δt = temperature of the test section thermocouple minus the temperature drop to the inside of the tube minus the interpolated value of the temperature of the air at that position assuming the specific heat constant, or:

$\Delta t = 317.5 - 0.2 - 87.2 = 230.1\ ^{\circ}F$

$$h = \frac{(1198)(12)}{\pi(0.524)(230.1)} = 38 \text{ BTU/hr ft}^2\ ^{\circ}F$$

5. Calculations of dimentionless parameters.

$$Re = dG/u = \frac{d(\dot{m}/A)}{u} = \frac{d\dot{v}\rho}{\frac{\pi D^2}{4}u} = \frac{4\dot{v}\rho}{\pi Du} = \frac{4 \times 12 \times 60}{\pi \times 0.524} \times \frac{\dot{v}\rho}{u}$$

$$= 1750 \times \frac{\dot{v}\rho}{u}$$

$\rho(87.2\ ^{\circ}F) = 0.0727$ lbm/cu.ft. ...(7)

$u\ (87.2\ ^{\circ}F) = 0.0452$ lbm/hr ft

$k\ (87.2\ ^{\circ}F) = 0.0153$ BTU/hr ft $^{\circ}F$

$Pr\ (87.2) = .7115$

$(Pr)^{0.4} = 0.873$

$$Re = 1750 \times \frac{10.6 \times 0.0727}{0.0452} \approx 33{,}400$$

$$Nu = \frac{hd}{k} = \frac{38 \times 0.524/12}{0.0153} = 108.3$$

$$\frac{Nu}{(Pr)^{0.4}} = \frac{108.3}{0.873} = 124.2$$


T/C LEAD HOLE
1/32 DRILL
4"
5"
2"
1.05"
0.524"
4 HOLES
13/64 DRILL
FLANGE
0.10"
60°
5/32"
2/3"
1/4 R
1 1/8"
7/8"
2"
12 3/8"
1 1/2"
3"
3"
3"
1 1/2"
0.524"
1.05"
1/16"
3/16"
4 HOLES #56 DRILL
0.200" DEEP
1/8"
TEST SECTION
6"
3 1/4"
1 3/4"
0.524"
1.05"
DRILL AND TAP FOR 1/8 NPS
COMPRESSION FITTING
3/4 NPS 14
END PIPE

FIG. VIII

FIG VII – TEST SECTION

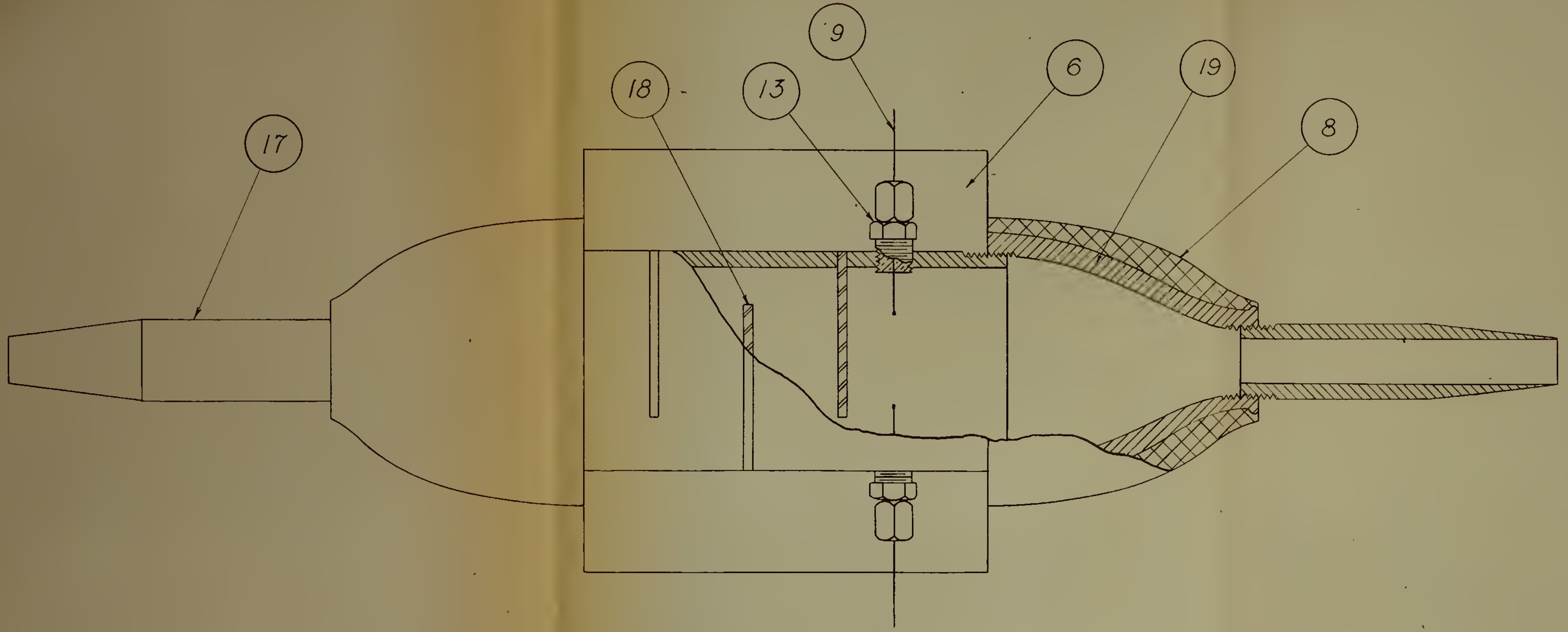


Fig. IX - Mixing Chamber

APPENDIX III

BILL OF MATERIALS

Part No.	No. Req'd	Name of Part	Specifications
1	1	Hot Pipe	Copper tube, 3/4" double extra heavy, seamless, cold-drawn
2	2	Flange	Bakelite
3	2	End Pipe	Same as part No. 1
4	1	Heating Element	#17 glass fibre insulated Nichrome V wire
5	2	Wire	#14 copper wire
6	1	Insulation	85% magnesia
7	2	Insulation	1/32" asbestos sheet
8	1	Tape	1" asbestos cloth tape
9	14	Thermocouple	#30 iron-constantan duplex
10	2	O-Ring	11/16" I.D. silicone rubber
11	1	Cover	Aluminum sheet 14 x 16
12	2	Strap	Aluminum sheet 1 1/4 x 17
13	4	Compression Fitting	1/8" NPS
14	4	Bolt	3/16" brass, length 14 1/4"
15	4	Insulation	Ceramic bead
16	6	Machine Screw	#8 round head, length 1/2"
17	2	End Pipe	Std. 1/2" pipe
18	3	Baffle	0.2" steel sheet
19	2	Bell Reducer	2" to 1/2" bell reducer, cast iron

Holden

The effect of vibration on heat transfer by forced convection to a compressible fluid in a horizontal tube.

26125

Thesis
H68

Holden

The effect of vibration on heat transfer by forced convection to a compressible fluid in a horizontal tube.